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Title: THERMAL CONTROL OF FLOWRATE IN ENGINE COOLANT SYSTEM

[001] This invention relates to coolant pumps, especially for automotive internal-combustion engines. The invention is aimed at providing a coolant pump which efficiently delivers flow characteristics in accordance with engine demand.

BACKGROUND

[002] In the traditional cooling systems, the coolant temperature might be several degrees from the optimum. Also, the coolant pump could draw far more energy from the engine than was required. The system has to provide enough cooling under the worst thermal load conditions (e.g a fully-laden vehicle ascending a steep grade on a hot day) and at the same time must not overcool the coolant and engine at the other extreme. Because of the compromises required to make the cooling system function at the extreme thermal conditions, during part-load conditions (as encountered most of the time) the coolant is not at its optimal temperature by several degrees, and the coolant pump wastes large amounts of energy.

[003] Patent publications EP-0,886,731 (30 December 1998) and US-6,499,963 (31 December 2002) disclosed engine cooling systems, in which the flowrate of coolant around the circuit was varied in accordance with the temperature of the coolant. It was disclosed that where the coolant pump was driven by e.g an electric motor, the pump speed could be kept constant, and flowrate could be made to vary solely in response to changes in coolant temperature; where the coolant pump was e.g engine driven, the flowrate could be made to vary in response both to changes in the coolant temperature and to changes in the engine speed.

[004] As disclosed, coolant passing through the pump rotor passed also through a set of movable swirl-vanes. The coolant flowrate was made to vary in response to changes in coolant temperature by providing that the orientation of the swirl-vanes was adjusted in response to changes in coolant temperature. As disclosed, the orientation of the swirl-vanes was varied from a position of boosting the flowrate to a position of inhibiting the flowrate, progressively, as a function of coolant temperature.

[005] One benefit of using the orientatable swirl-vanes to control coolant flowrate is that a designer can so design the system that the amount of energy needed to drive the pump is

(almost) proportional to flowrate. This may be contrasted with cooling systems in which the flowrate is controlled by e.g throttling the flow from the pump, in which case the energy drawn by the pump remains high even when the flowrate is small. It also may be contrasted with systems in which the flowrate has been controlled by e.g varying the speed of the pump rotor, when it can be difficult to engineer a pump to have reasonable efficiencies over a large range of rotor speeds.

[006] One benefit that can arise in a system in which flowrate is adjusted in accordance with coolant temperature is that the coolant temperature can be kept constant, during operation of thè engine, within quite close limits. It is not unrealistic for the designer now to aim to keep the temperature constant (once the coolant has warmed up) over the whole range of engine speeds, loads, ambient temperatures, and other relevant operating conditions, to within plus/minus two C-degrees. (It should be noted, in the traditional automotive cooling system, that the (warmed-up) temperature can vary plus/minus five, or even ten, C-degrees over the range of conditions.)

[007] Engine designers can take advantage of this constancy to make the engine operate more economically; in particular, engine performance and efficiency are often highly dependent upon engine oil temperature, and when that is kept constant (as it tends to be when the coolant temperature is kept constant) over a long running period the improvement in fuel consumption can be substantial.

[008] Traditionally, automotive engines have included a (mechanical) thermostat, structurally based on a bulb containing an expandable wax, for managing coolant temperature by controlling flow to the radiator. Basically, the thermostat cuts off or reduces flow through the radiator when the coolant is below a certain temperature, and only allows full flow when the coolant in the engine has warmed up above that temperature. The above-mentioned patent disclosures, however, referred only to the thermal control of the coolant flowrate during normal running, i.e after the coolant has warmed up.

GENERAL ASPECTS OF THE INVENTION

[009] The present invention relates to combining the temperature-controlled swirl-vanes technology with the requirement an engine has for thermostat control of coolant flow to the radiator. In the above-mentioned disclosures, it was arranged that the temperature-controlled swirl-vanes unit was provided as a structurally separate component of the vehicle

from the warm-up thermostat. As will now be described, both the function of modulating the coolant flowrate in accordance with coolant temperature (using thermostatically controlled swirl-vanes), and the function of blocking the coolant flow from passing through the radiator in accordance with coolant temperature, can be provided in a common structure.

[0010] The structure that blocks flow through the radiator when the coolant is cold can be regarded as comprising a radiator-port, and a radiator-port-closer. The radiator-port-closer is moved from a closed or blocking position to an open position by means of a rad-port-thermal-unit, which is arranged such that the radiator-port is closed when the coolant is cold, and is open when the coolant has warmed to running temperature. Traditionally, this function is carried out by the traditional mechanical wax-type thermostat structure, but other structures have been arranged to have the equivalent function.

[0011] A benefit that arises from housing both the radiator-port-closer and the swirl-vanes within the pumping-chamber is that the same structure that is highly suitable for carrying out one of those functions is also highly suitable for carrying out the other. The pump housing, containing the pumping-chamber, is a structure that is aimed at controlling and adjusting the flowrate, and controlling and adjusting the direction of flow, of coolant as it enters the impeller. The pump housing is designed to do that. The economies arise from the fact that the same structure is eminently and economically suitable and available also to serve as the structure that directs and controls the flowrate of coolant passing through the radiator.

[0012] Another benefit that arises from housing both the radiator-port-closer and the swirl-vanes within the pumping-chamber is improved pumping efficiency. Of course, it is all too possible to design a pump (i.e any pump) to be inefficient: disruptions to the flow can be caused by the flowpath-defining walls being irregular, and especially if the walls are so shaped that the flowing liquid has to accelerate and decelerate repeatedly. The designer's aim should be to so arrange the flowpath walls that the coolant liquid flows steadily and smoothly. The aim should be to minimise the number, and the abruptness, of changes in velocity - that is, of changes in cross-sectional area. Inside the engine (and inside the radiator) the many flowpaths are small and narrow, in order to maximise the rate at which heat is transferred into and out of the coolant liquid. The aggregate cross-sectional areas of the many narrow flowpaths is relatively large, whereby coolant tends to flow relatively slowly through the engine and radiator. But, within the pipes and hoses and other portions of the flowpath that convey coolant directly to and from the pump, the flow is constrained rather in a single conduit, of a cross-sectional area that is relatively small compared with the aggregate area of the many flowpaths inside the engine and radiator. Hence, the zone of

the coolant circuit in which the velocity of the coolant is highest tends to be the conduits leading into and out of the engine and radiator, and particular in those conduits that lead to and from the pump.

[0013] Thus, it is particularly in these zones of higher coolant velocity, in the conduits outside the engine and radiator, that the designer should seek to avoid abrupt, large, changes in cross-sectional area of the flowpath. It is recognised that this aim can be addressed most easily when both the radiator-port-closer and the swirl-vanes are housed, together, inside the pumping-chamber. This juxtaposition enables the overall resistance of the flowpath to be minimised. Of course, it is possible, even when the radiator-port-closer and the swirl-vanes are housed together inside the pumping-chamber, for the pumping-chamber to contain abrupt changes in cross-section which spoil and disrupt the smooth flow through the pump; the point is that housing both components inside the pumping-chamber maximises the opportunity to provide a smooth steady progression of the flow velocity as the coolant approaches and leaves the impeller.

[0014] Other economies and efficiencies arise from the fact of housing both the radiator-port-closer and the swirl-vanes within the pumping-chamber. One preferred option is to use one single common thermal-drive-unit to drive both the swirl-vanes and the radiator-port-closer. Another preference is to use the swirl-vanes themselves not only to control the flowrate during normal running, but also to close off the radiator-port. In that case, the designer may provide two separate thermostats (or equivalent thermal-drive-units) to drive the swirl-vanes over different parts of the temperature range; but preferably, for greatest economies, not only are the swirl-vanes and the radiator-port-closer provided as one single mechanically-unitary structure, but also the whole range of movement of that structure is driven by one mechanically-unitary thermal-drive-unit.

DESCRIPTION OF PREFERRED EMBODIMENT

[0015] The invention will now be further described with reference to the accompanying drawings, in which:-

Fig 1 is a sectioned plan view of a coolant pump for an automotive application, the section being taken at the level of the swirl-vanes, showing inlet ports for conveying coolant into the pump from the radiator and from the engine/heater of the vehicle.

Fig 2 is a section of the same pump at the level of the impeller rotor, showing the outlet port for conveying coolant from the pump, back into the engine.

Fig 3 is a section of the same pump at the level of a thermostat actuator.

Fig 4a is a diagram of the pump showing the swirl-vanes at a full-closed position.

Fig 4b shows the swirl-vanes orientated to an almost-full-closed position.

Figs 4c,4d,4e show the swirl-vanes opening in progressive degrees.

Fig 4f shows the swirl-vanes orienated to an almost-fully-open position;

Fig 5a is a section of view of another coolant pump;

Fig 5b is the same section as Fig 15a, but shows the pump in a different condition;

Fig 5c is the same section as Fig 15a, but shows the pump in another different condition.

Fig 6 is a block diagram showing some of the components of a typical coolant circulation system.

Fig 7 is a cross-sectioned elevation of the coolant pump of Fig 1.

Fig 8a is a portion of a view similar to Fig 17 of another pump, having a dual impeller;

Fig 8b is the same view as Fig 8a, but illustrates a different condition.

Fig 9 is a cross-section of another coolant pump.

Fig 10 is a pictorial partly-sectioned view of the pump of Fig 9.

Fig 11a is a diagram illustrating an operating condition of a coolant pump similar to that shown in Fig 9;

Fig 11b is the same diagram as Fig 11a, except that the pump is in a different operating condition;

Fig 11c is the same diagram as Fig 11a, except that the pump is in another different operating condition.

Fig 12 is a graph showing a mode of operation of a thermostat unit that is suitable for use in the invention.

[0016] The apparatuses shown in the accompanying drawings and described below are examples which embody the invention. It should be noted that the scope of the invention is not necessarily defined by specific features of exemplary embodiments.

[0017] In the coolant circulation pump mechanism 230 of Fig 1, a rotating vanes-ring 232 carries a set of swirl-vanes 234. In this pump, coolant enters the impeller 236 from two sources, being the radiator-port 237 and the engine/heater by-pass port 238. The flow from the ports 237,238 passes through the swirl-vanes 234, before entering the blades of the impeller 236.

[0018] The swirl-vanes 234 are operated on by the vanes-ring 232. The vanes-ring 232 is rotatable, its orientation being under the control of a thermostat unit 235. (In alternative embodiments, other types of thermally-controlled actuator may be used in place of the

thermostat 235.)

[0019] A drive-pin 239 connects the stem of the thermostat 235 with the vanes-ring 232. When the stem moves, the drive-pin 239 causes the vanes-ring 232 to rotate in a movement that corresponds to, and is in unison with, the movement of the stem. The swirl-vanes 234 are carried in respective pivots mounted in the housing of the pump, whereby the rotation of the vanes-ring 232 causes the angle or orientation of the swirl-vanes to change.

[0020] Fig 4a shows the components of the pump 230 in the COLD position, being the position they adopt while the coolant entering the pump through the heater port 238 is cold (i.e not yet warmed up). In this COLD position, coolant cannot pass from the radiator port 237 into the impeller, because the swirl-vanes 234 lie orientated to the closed position. It is probably unavoidable that there will be some slight leakage through the vanes when the vanes are closed; however, it is recognised that the resulting small radiator flow can be tolerated in most applications.

[0021] Fig 4d shows the swirl-vanes in their WARM orientation. Here, the swirl-vanes are slightly opened. The coolant has warmed up sufficiently that the coolant needs to be cooled, by being passed through the radiator, but the coolant is at the low end of this warm-hot range. Now, the flowrate of coolant needs to be much less than the flowrate when the coolant has risen to the upper end of its (allowed) range of temperature. The swirl-vanes reflect this requirement, in that the swirl-vanes are orientated to provide less flow boost (i.e to provide flow reduction) in Fig 4d than in Figs 4e and 4f. On the other hand, in Fig 4d (WARM) the flowrate is nowhere near zero, whereas the flowrate does approach zero in Fig 4a (COLD).

[0022] Attention is directed to the above-mentioned publications for a detailed description of the manner in which the swirl-vanes boost the flowrate through the pump when the swirl-vanes direct the flow to swirl in the opposite directional sense to the rotation of the impeller; when the swirl-vanes direct the flow to swirl in the same directional sense as the impeller, the flowrate is reduced. The effect is progressive; i.e as the coolant goes from WARM to HOT, and as the swirl-vanes move from their maximum-same-sense orientation to their maximum-opposite-sense orientation, the flowrate through the impeller increases more or less linearly, from maximum flow-reduction to maximum flow-boost, in proportion to the change in the angle of orientation of the vanes. Fig 4a shows the swirl-vanes in their COLD, fully closed, position, which is an embodiment of the present invention, which of course is not described in the said publications. Figs 4a-4e show the swirl-vanes opening progressively from the fully closed position (Fig 14a), through their WARM position (Fig 4d) in which the swirl-vanes are

biassing the flow WITH the direction of rotation of the impeller, to their HOT position (Fig 4f) in which the swirl-vanes are biassing the flow AGAINST the direction of rotation of the impeller.

[0023] It should be noted that, under some cold climate conditions, a traditional thermostat rarely opens at all, i.e the coolant hardly warms up above the temperature at which coolant just starts to pass through the radiator. The engine in that case is overcooled, with resulting poor fuel consumption, increased emissions, and possibly reduced durability of engine components due to non-optimal oil temperature. Of course, these measures are forced on the designer, who has to design the system as a series of compromises, in order to cater for other climate conditions. It is recognised that the elimination of the traditional thermostat, i.e the combining of the thermostat function into the swirl-vanes, as described, makes it much easier for the designer to alleviate the traditional compromises, and thus to arrange for the coolant to achieve its optimum operating temperature over a wider range of climate conditions.

[0024] Figs 5a,5b,5c show a modified arrangement, having just a single swirl-vane 240. (It will be understood that the expression "a set" of swirl-vanes, as used herein, reads onto just one swirl-vane, where that is the case.) Here, when the coolant is cold, the swirl-vane 240 blocks coolant from the radiator-port from reaching the impeller. When the coolant is warmed up (Fig 5b), coolant can enter the impeller from both ports.

[0025] It may be appropriate, in some cooling systems, for the designer to arrange for the input from the engine/heater by-pass to be completely blocked, and this can be done if required (Fig 5c). It will be noted that in Fig 5a, when the coolant is cold, the swirl-vane is directing the flow from the engine/heater by-pass port against the direction of rotation of the impeller, which boosts the flowrate; whereas the flow from the radiator port (Figs 5b,5c) is directed in the same rotational sense as the impeller, which reduces the flowrate-boost in that case.

[0026] In Figs 5a,5b,5c the swirl-vane 240 is driven to rotate, not directly by a wax-bulb type of thermostat element, but by an electric-motor/gearbox arrangement 241. The motor is a stepper-motor, and its rotational position is controlled by signals from a temperature sensor located at a suitable point in the coolant circuit, which may be mechanically separate from the motor/gearbox 241. It should be understood that the motor/gearbox arrangement used in Figs 5a,5b,5c, with its separate temperature sensor, could be used in place of the mechanical thermostat unit of Fig 1, and vice versa. A thermostat (which combines thermal sensor and actuator in one mechanical unit) is not so sophisticated and versatile as to its

functionality, but is more economical. Other kinds of thermostat unit may be used, for example bi-metallic units.

[0027] In Fig 1 and in Figs 5a,5b,5c the illustrated structures provide mechanical coordination between the swirl-vanes orientation mechanism, including the vanes-rings 232, and the valve-member orientation mechanism, including the drive-pin 239 or the motor/gearbox 241.

[0028] The cooling system of which the pump of Fig 1 is a component is of the type in which coolant circulates at all times through the heater (Fig 6). (In other types of cooling system, flow may be sometimes, in operation, diverted to by-pass the heater.) In Fig 6, the impeller of the pump P is driven e.g by means of a geared drive, or by means of a belt drive 241, directly from the engine E. In Fig 6, when the coolant is warmed up, the coolant circulates around the radiator R; when the coolant is cold, coolant cannot circulate around the radiator R, because the swirl-vanes 234 in the pump P lie in a fully-closed position, thus closing off the radiator-port 237. The temperature-sensing bulb in the thermostat-unit 235 is positioned appropriately to measure the temperature of the coolant coming from the engine E (and, or via, the heater H) just before the coolant enters the pump P. As shown in Fig 1, there is a passage 248 between the heater port 238 and the bulb, whereby the bulb is flooded with incoming coolant.

[0029] It will be noted that the separate thermostat, which automotive engines usually have, has been eliminated in the circuit of Fig 6.

[0030] There are many different configurations of the components of automotive cooling systems, and the designer will arrange the pump inlets/outlets to suit. That is to say: linking the radiator shut-off thermal control with the swirl-vanes thermal control, as described, might or will require different configurations with different engine systems.

[0031] In Fig 1, the swirl-vanes are in their HOT position - the coolant having warmed up - whereby coolant enters the coolant circulation pump 230 both from the heater-port 238 and from the radiator port 237. The mouths of the ports 237,238 are arranged such that coolant passing into the pump from the heater-port 238 passes straight into the impeller, whereas coolant from the radiator-port 237 passes through the swirl-vanes 234.

[0032] When the coolant passing through the pump 230 is cold, i.e has not yet warmed up, it is desired that the radiator be closed off from the circulating coolant. This is shown in

Fig 4a, where flow from the radiator is blocked off, in that the swirl-vanes 234 lie orientated in such position as to block flow from the radiator port 237, i.e to prevent coolant from the radiator from passing through to the impeller 236. The swirl-vanes have been driven to this position by the thermostat-unit 235 which, in Fig 4a, is in its fully-retracted, COLD, condition. Thus, when the coolant is cold, the coolant passing through the pump, and entering the engine, comprises only coolant that has just come from the engine, via the heater; coolant from the radiator cannot enter the pump, and cannot enter the engine, because the vanes 234 are closed.

[0033] As the coolant circulating around just the engine (and heater) warms up, so the bulb of the thermostat-unit 235 expands, which drives the vanes-ring 232 in an anti-clockwise direction, causing the vanes 234 to open. Now, coolant from the radiator can pass through to the impeller 236.

[0034] After that, once the coolant has warmed up, the temperature of the coolant varies in accordance with driving conditions, vehicle loading, ambient temperature, etc; as the coolant becomes hotter, or becomes less hot, the swirl-vanes vary as to their orientation, in accordance with the coolant temperature, in the manner as described in the publications. Again, the designer should arrange that, once the coolant is up to normal running temperature, the angle the swirl-vanes 234 adopt when the coolant is at its hottest gives the greatest boost to the flowrate, whereas the angle the vanes adopt when the coolant is at the cooler end of its range of normal-running temperatures gives the greatest reduction (or, it may be termed, gives the smallest boost) to the normal-running flowrate. Typically, the minimum normal-running flowrate may be of the order of a half of the maximum normal-running flowrate, at a typical pump speed and operating condition. In Fig 1 the impeller 136 rotates in an anti-clockwise direction, whereby the above manner of operation obtains.

[0035] Attention is drawn to the following point that arises from moving the thermally-operated radiator-port-closer from its traditional separate thermostat housing into the pumping-chamber, and by combining the radiator-port-closer with the swirl-vanes. One of the banes facing designers of automotive cooling systems is the inordinately high flowrate-resistance attributable to the traditional conventional thermostat, even when in its fully open condition. Theoretically, the problem of the high pressure drop across the traditional thermostat might be solved by arranging the passageways to avoid the abrupt changes in cross-section that characterise the flowpath through the thermostat. But redesigning the thermostat and its housing to achieve that desideratum, without compromising other aspects of performance, has proved difficult in practice, and designers have had to accommodate the

large pressure drop through the thermostat, i.e through the radiator-port-closer.

[0036] Combining the radiator-port-closer function into the structure of the swirl-vanes, however, removes or reduces the problem of the large pressure drop. Now, effectively, the open-thermostat flow resistance is the same as the maximum flow-boost condition of the swirl-vanes, i.e the condition as illustrated in Fig 1. Now, there is no flow resistance at all, when the swirl-vanes are at the maximum-boost orientation, to compare with the large resistance associated with the traditional thermostat.

[0037] It will be noted from Fig 1 that the swirl-vanes 234 (numbering thirteen swirl-vanes in this case) do not completely surround the impeller 236. A sector of the circumference of the impeller is left open, being the sector communicating with the engine/heater inlet port 238, i.e the flow that by-passes the radiator during warm-up. Thus, even when the swirl-vanes are fully closed (Fig 4a), it is only the radiator port 237 that is blocked, not the by-pass port 238. When the coolant is cold enough for the radiator to be blocked the flowrate passing through the engine is quite small, which is reflected by the fact that this flow occupies only a small sector 233 of the circumference of the intake of the impeller. The full HOT flowrate passing through the radiator will be many times greater than the low flowrate of the by-pass flow passing through just the engine/heater in the COLD condition.

[0038] The swirl-vanes are most effective when they are arranged to completely, or almost completely, surround the intake of the impeller. If some of the flow entering the impeller has not been through the swirl-vanes, then the flowrate is not being fully and completely controlled responsively to the swirl-vanes, i.e responsively to the temperature-dependent orientation of the swirl-vanes. Preferably, the designer should see to it that as much as possible of the warmed-up flow of coolant passes through the swirl-vanes. In other words, the sector 233 of the impeller circumference that receives incoming flow from the engine, during warm-up from cold, should be minimal. The full flowrate from the radiator under HOT conditions preferably should occupy eighty or ninety percent of the circumference of the intake to the pump impeller; and should occupy at least about sixty percent of the circumference, as a minimum.

[0039] In some cooling systems, it is possible to arrange for the swirl-vanes to occupy the whole circumference of the intake of the impeller, and that is best from the standpoint of thermally-responsive control of the flowrate. However, it is recognised that the loss of swirl control over a small sector is not significantly detrimental to swirl effectiveness.

[0040] In some engines, the designer may choose to block off flow through the heater core until the coolant has warmed up. Alternatively, the designer may even choose to block flow around the engine until the coolant has warmed up. While this latter makes for the most rapid warm-up, special attention must be paid to sensing the coolant temperature in the engine, and it may be necessary to sense the temperature in the cylinder head, close to the exhaust valves, which is likely to be the hottest area, and which is inevitably spaced some distance from the coolant pump. The designer may prefer, then, that measuring the temperature be done by electronic thermal sensors, the data signals from which are analysed, and used to operate e.g a servo to actually effect the movement of the swirl-vanes.

[0041] Where a conventional wax-bulb type of the thermostat bulb is used, the bulb preferably should be wetted by coolant coming from the engine/heater by-pass circuit, as in Fig 1.

[0042] Fig 7 is a cross-section of the pump 230 of Fig 1. The pump impeller 236 is driven, in this case, by means of a drive belt from the engine, which operates on a drive-pulley 243. Thus, the speed of the pump varies in direct proportion to the speed of the engine. Driving the coolant pump from the engine, although that is a traditional and very common technology, poses the problem that at low engine speeds the pump output (i.e the litres per minute of coolant flow produced by the pump) might not be enough to remove all the heat the engine puts into the coolant; equally, at high engine speeds, the flowrate is much larger than required, which results in engine power being wasted, and indirectly in that the cooling system has to be engineered to cope with the high flowrates and/or pressures. The heater often has a relatively high flow resistance; so, when the coolant is cold, and the radiator is not in circuit, there is need for the pump to produce a higher pressure, which can pose an additional problem at low engine speeds.

[0043] The designer is thus faced with a compromise, in that the impeller has to produce an adequate flowrate and pressure at low pump speeds, and yet must not produce excessive flowrates and pressures at higher pump speeds. The need for compromise is exacerbated in that, when the coolant is cold but the heater is in circuit, although the flowrate then is low, the extra resistance of the heater imposes the need for that low flowrate to be produced at a higher pressure. One approach to easing this compromise is to provide the impeller with two sets of blades, and to engineer the impeller such that at low speeds (i.e low flowrates) both sets of blades are available to pump the coolant, whereas at high pump speeds (i.e high flowrates) one of the sets of blades is by-passed. The pump impeller 236 has two sets of blades, with the effect as shown in Figs 8a,8b.

[0044] The impeller 236 includes a set of primary (mixed axial and radial flow) blades 244 and a set of secondary (radial) blades 245. When the pump drive speed is low, and the flowrate is low, the coolant passes axially through the primary blades 244; the pumped liquid then changes direction, and passes around the promontory 246, and thence passes into the entrances of the secondary blades 245, and then radially through the secondary blades (Fig 8a), generating the desired higher pressure.

[0045] On the other hand, when the impeller speed is high, the flow from the primary blades 244 now has so much axial-velocity momentum that the coolant tends to by-pass the entrances of the secondary blades 245 (Fig 18b). Thus, the secondary blades become starved of liquid.

[0046] The secondary blades 245 are radial, whereby the pressure differential between the entrances and the exits of the blades 245 is created by centrifugal force, and can be quite substantial. Thus, provided the liquid near the promontory 246 is moving slowly, the liquid is drawn, quite strongly, into and through the secondary blades 245. It is recognised that the flow route or pathway around the promontory 246 can be made so tortuous that, as mentioned, at higher speeds, only a smaller proportion of the axial flow emerging from the primary blades 244 reaches the secondary blades 245.

[0047] Thus, at low pump-speeds, a high percentage of the flow passes through both the primary blades 244 and the secondary blades 245, whereas, at high pump-speeds, only a much lower percentage of the flow passes through both the primary blades 244 and the secondary blades 245, in that, at high pump speeds, most of the flow passes straight into the outlet scroll chamber 247 without passing through the secondary blades.

[0048] The effect is that the ability to overcome the relatively higher heater circuit resistance is boosted at low speed because then most of the flow passes through both sets of blades; whereas, at higher speeds, most of the flow by-passes the secondary blades.

[0049] Fig 9 shows another structure in which a vanes-orientation mechanism is mechanically coordinated with a radiator-port-closing mechanism. Fig 10 shows the same structure pictorially, partly in cross-section.

[0050] In Fig 9, coolant from the automobile's radiator enters the pump chamber 254 via radiator-port 256. Located in the chamber is a slider 257. When the coolant is hot, the slider 257 lies towards the rightwards extreme, as shown in the lower half of Fig 9.

[0051] The open interior conduit 258 of the slider 257 has a radially-outwards-facing opening 259. This opening 259 connects with the radiator-port 256 when the slider 257 is to the right. Coolant enters the pumping-chamber 254 from the radiator, and passes to the pump impeller 260. The radiator-port 256 is blocked when the coolant is cold (upper-half of Fig 9) and open when the coolant has warmed up (lower-half of Fig 9.

[0052] Before reaching the blades of the pump impeller 260, the coolant from the radiator-port 256 passes through the swirl-vanes 262. The swirl-vanes 262 impose a bias to the flowing coolant, giving the coolant a rotary swirl motion. Depending upon the orientation of the swirl-vanes, this swirl motion can be in either the same rotational sense as the rotation of the impeller, or the opposite sense. Again, when the swirl-vanes are orientated AGAINST the rotation of the impeller, the volumetric flowrate and pressure through the impeller are boosted, whereas when the swirl-vanes are orientated WITH the rotation of the impeller, the flowrate and pressure are reduced. The swirl-vanes are orientatable progressively, from a maximum flow-boost orientation through a maximum flow-reduce (or minimum flow-boost) orientation.

[0053] The swirl-vanes 262 are mounted in a vane-mounting-structure, comprising a cage, which comprises an inner ring 264 and an outer ring 265. The two rings are fixed together, to form the cage. The two rings define an annular passageway 267. The swirl-vanes straddle the annular passageway 267, radially between the two rings 264,265.

[0054] The rings 264,265 carry respective pivot bearings 268,269, in which the swirl-vanes 262 are rotatably mounted. The pivot pin 270 of the swirl-vane 262 has an extension 272, which extends through the bearing 269 in the outer ring 265, and a lever arm 273 is carried on the extension 272. The orientation of the swirl-vane 262 is adjusted by moving the lever arm 273.

[0055] The cage 263 is carried in the fixed chamber 254. A peg (not shown) engages a socket in a shoulder 274 of the chamber, to constrain the cage 263 against rotation within the chamber.

[0056] A spring (not shown) serves to urge the lever-arms 273 of the swirl-vanes 262 to the left. Noting the direction of rotation of the pump impeller 260, the designer arranges the apparatus so that the more the lever-arms 273 lie to the left (in Fig 9), the more the swirl-vanes 262 are orientated to the flow-reducing condition. As the lever-arms 273 are moved to the right, the swirl-vanes 262 become more orientated towards the flow-boosting condition.

The design of the lever arm and the slider geometry can be designed to suit the particular desired relationship of swirl-bias to slider motion.

[0057] Inside the pump chamber 254 is a thermostat unit 275. The unit 275 is conventional, in itself, and includes a bulb which expands as it heats, driving a stem 276 out of the thermostat casing 278. The casing is a press fit inside the slider 257. (Again, it will be understood that a thermally-controlled movement-actuator other than a traditional wax-type thermostat may be provided, e.g an electrical linear actuator coupled to a thermal sensor, for the purpose of moving the slider.)

[0058] As the stem 276 moves out of the casing 278, due to a rise in temperature of the coolant flowing over the casing 278, the casing, and the slider 257 to which it is attached, move to the right. The nose 279 of the slider 257 engages the lever-arms 273, whereby thermally-induced movement of the slider in the left-right sense moves the lever-arms 273, giving rise to a change in the orientation of the swirl-vanes.

[0059] A lost motion provision may be incorporated into the Fig 9 design. The designer can provide a gap 281 between the nose 279 and the lever-arms 273. The larger the gap 281, the greater the lost motion, as the coolant warms, before the lever-arms 273 move. The lost motion provision can be coordinated with the point at which the radiator-port 256 opens.

[0060] Designs based on the Fig 9 illustration can be highly suited to automotive use. The pump unit is structured as a mechanically-compact unit, which can be designed to be attached to the engine-block on a simple bolt-on basis. The unit is self-contained, in that it can be assembled and tested, for most of its functions, while off the engine. In an alternative design, the pump unit is housed within the engine block, rather than in a separate bolt-on housing.

[0061] It may especially be noted that the slider 257 and the cage 263 are both accommodated inside the smooth-bored interior of the pump chamber 254. Thus, for servicing, both the slider and the cage can be simply slid out of the chamber, upon removal of the end-cover 277, and this can be done without removing the unit, and without disturbing the hose connections. As mentioned, the cage 263 is pegged against rotation relative to the chamber, and it does not matter if the slider 257 should tend to rotate.

[0062] Other arrangements of the components may be engineered: for example, it may be arranged that the cage slides with the slider, whereupon the lever arms may be caused to

rotate by contacting the shoulder 274. The thermostat unit may be attached to the end cover, rather than to the slider; however, the designer should prefer an arrangement in which the temperature-sensing portion of the thermostat is actually immersed in the flowing coolant.

[0063] As discussed in relation to the embodiment of Fig 1, it is desirable for the thermally-orientated swirl-vanes to affect as much as possible of the flow entering the intake of the impeller. In that case, a small (insignificant) sector of the circumference of the intake was left uncontrolled by the swirl-vanes, to enable the swirl-vanes to serve as the means for blocking the radiator-port before the coolant has warmed up from cold. In Fig 9, as described, the swirl-vanes do not serve as the means for blocking the radiator-port, and therefore the swirl-vanes can occupy the whole of the cross-sectional area of the intake into the impeller blades.

[0064] In the embodiment of Figs 9,10 it should be noted that, as in the previous embodiment, the swirl-vanes are juxtaposed close to the radiator-port and associated radiator-port-closer. This makes for a compact, economical assembly. The juxtaposition also means that the coolant flow can approach very closely to the ideal of a smoothly progressive reduction in cross-sectional area of the coolant as it enters and passes through the impeller, whereby the resulting change in velocity is also smoothly progressive, and the resulting losses due to flow interruptions are minimised.

[0065] Comparing the Fig 1 embodiment with the Figs 9,10 embodiment, in both cases the swirl-vanes are pitched evenly around a pitch-circle, which is concentric with the rotary axis of the impeller. In the latter embodiment, the swirl-vanes are positioned axially in-line with the impeller, at a place where the coolant is moving axially towards the intake of the impeller, and the swirl-vane pivots lie on axes that are radial with respect to the impeller axis. In the former embodiment, the swirl-vanes lie around the impeller, at a place where the coolant is moving radially inwards into the intake of the impeller, and the vane pivots lie on axes that are parallel to the impeller axis. The latter embodiment disposes the incoming coolant in what may be regarded as a flat spiral around the intake, whereas the former embodiment disposes the incoming coolant in what may be regarded as a cylindrical tube that is co-axial with the impeller. The designer may choose the embodiment in accordance with the available space: if there is more space for the flow-control apparatus to protrude axially rather than radially, the latter embodiment will be preferred; if axial space is more critical, the former would be preferred.

[0066] Another example of a manner of coordinating coolant flows in the circulation circuit will now be described, referring to Figs 11a,11b,11c.

[0067] When the coolant is cold, in a traditional automotive coolant circulation system, the thermostat has prevented coolant from passing through the radiator. When the coolant nears its normal running temperature, the thermostat opens, only then permitting flow through the radiator. However, in traditional automotive systems, the cold coolant, though cut off from the radiator by the closed thermostat, still flows through the heater circuit.

[0068] In traditional heater circuits, all or part of the coolant flow that is routed around the engine is also routed around the heater circuit. Some heater circuits include a manually-operated valve, which shuts off flow through the heater, effectively diverting a greater proportion of the coolant flow through the engine by-pass or radiator circuit - i.e not through the heater - thus controlling the heat output of the heater.

[0069] Often, when the vehicle is starting from cold, on a cold day, the driver turns the heater control to full heat. If so, a significant portion of the coolant, as it flows around the engine, also flows through the heater, and this can delay warm-up of the coolant in the engine. Delayed warm-up is not preferred, not just for the heater, but especially from the standpoint of engine wear. The time for warm-up can be improved if the heater is kept out of circuit until the coolant is at least partially warmed up. The driver cannot gain any benefit from the heater, anyway, until the coolant has warmed up.

[0070] Cutting off flow to the heater when the coolant is very cold, in a traditional system, would seem to require a separate thermostat, because the temperature at which flow should be admitted to the heater is different from the temperature at which flow should be admitted to the radiator.

[0071] When the radiator thermostat, i.e the mechanism for opening /closing the radiator port, is mechanically coordinated with the mechanism for changing the orientation of the swirl-vanes, as described herein, it is recognised that it is hardly any further complication to arrange for the mechanism also to open /close the heater port, and to do so at the required different temperature.

[0072] Figs 11a,11b,11c show how this may be done. Coolant from the heater enters via heater port 283, and coolant from the radiator enters via radiator-port 284. The coolant is conveyed along the conduit 285 in the slider 286 to the swirl-vanes, which lie to the right, as in Fig 9. The slider 286 moves responsively to a temperature-sensitive actuator (not shown).

[0073] Fig 11a shows the situation when the coolant is very cold. Here, both the heater-port

283 and the radiator-port 284 are closed, whereby the coolant only circulates around the engine. Designers usually arrange that coolant can still circulate around the engine, even when flow through the heater circuit is closed: therefore, the heater by-pass conduit must have its own entrance port into the pumping-chamber, which must be separate from the heater-port 283 since the heater-port 283 may be closed. The by-pass entrance port is not shown in Figs 11a,11b,11c.

[0074] As the coolant starts to warm up, from very cold, the slider 286 moves to the right. Now, although the radiator-port 284 remains closed, the heater-port 283 is open, and the partially warmed-up coolant can circulate round the heater.

[0075] As the coolant approaches warmed-up running temperature, the radiator-port 284 also opens. Now, coolant can circulate through the heater and around the radiator.

[0076] As shown in Fig 11c, when the coolant is at the limit of maximum hotness, flow through the heater-port 283 is cut off, or is almost cut off.

[0077] Whether the heater port remains partly open or is completely closed at very hot temperatures, the point is that the mechanism as described makes it an easy matter for the designer to choose the opening /closing sequences. The exact nature of the overlap or non-overlap of the heater and radiator ports makes little difference to the cost or complexity of the apparatus, giving the designer freedom to arrange overlap as may be desired. The designer may wish to arrange that the flow can pass through the heater even when the coolant is very hot.

[0078] In Figs 11a,11b,11c the slider 286 also operates the mechanism for orientating the swirl-vanes, and the designer should ensure the correct correspondence and overlap between the closing /opening of the ports and the orientation of the vanes, which will secure good efficiency of the engine under a wide range of operating conditions. But again, the designer is free to choose the exact sequence of closing /opening of the heater and radiator ports, and their inter-relationship with the orientation of the swirl-vanes, i.e is free to choose in the sense that, whatever the chosen sequence, it makes little difference to the cost or complexity of the apparatus.

[0079] Some of the following variations in the system are also considered. For example, the coolant pump impeller (rotor) may be centrifugal (radial), or may be a propeller (axial), or a combination. As another example, the designer might prefer to provide a small

supplementary pump for the heater, rather than have the heater flow go through the main pump.

[0080] Another variation in the system concerns the orientatable swirl-vanes themselves. The designer should see to it that the swirl-vanes are able to be re-orientated, when that is needed, in a reliable trouble-free manner, over a long service life. However, pivot connections and sliding interfaces can lead to reliability problems. In an alternative structure, the swirl-vanes flex, rather than pivot. That is to say, the vanes are so structured as to bend, rather than pivot, in response to the thermal signal.

[0081] The efficiency of the pump assembly is measured as the product of the volumetric flowrate and the pressure rise of the pumped liquid, per watt of power needed to drive the pump. This efficiency is bound to vary, to an extent, with the degree of orientation of the swirl-vanes. It is recognised, however, that the efficiency of the pump in fact does not go down very much, as the swirl-vanes are re-orientated. It is recognised as a feature of the swirl-vane re-orientation system, as a structure for controlling flowrates through rotary pumps, that the efficiency (i.e the wattage from the motor or driver needed per unit of pressurised flow-rate) varies relatively little, over a wide range of flowrates, when compared to other flow control structures.

[0082] It is also recognised that the flowrate produced by the pump, as measured in litres per minute, is controllable over a wide range of flowrates, by controlling the orientation of the swirl-vanes.

[0083] By contrast, traditional flowrate-control systems have left the pump subject to large changes in efficiency at the different speeds. The pump would be designed for good efficiency at a particular operational flowrate, but the pump would be very inefficient at other speeds.

[0084] The changes in flow produced by the changes in orientation of the swirl-vanes can be done over a wide range, and without as significant a loss of efficiency over a wide range, as contrasted with other flow-control systems, for example systems in which a blocker moves to close off a port.

[0085] It is not required, in the invention, that there be only one thermal sensor. When the thermal sensor takes the form of a mechanical thermostat bulb unit, it can be difficult to coordinate more than one sensor; but when the thermal sensor provides an electronic signal,

which is fed onto the engine data bus, there is little difficulty in accommodating and coordinating several sensors, if the designer so wishes. For example, in some installations, the designer may prefer to have temperature sensors e.g at the pump intake, in the engine near the exhaust valves, in the radiator, in the heater, in the pump outlet, etc, and (especially) in the engine oil. Then, as engine operating conditions change, the orientations of the swirl-vanes may be coordinated in a more refined and sophisticated manner, aimed at optimising the operating temperature of the engine, and aimed at reducing deviations from the optimum as quickly as possible.

[0086] The bus data from the coolant temperature sensors can also be arranged to control the radiator fan, as well as controlling the swirl-vane orientation. For example, the designer may set the system such that, if there is not much temperature drop across the radiator, the fan may be switched on, or sped up, and coordinated with the orientation of the swirl-vanes.

[0087] As mentioned, the temperature sensor(s) may be electronic, and provide simply a voltage, or simply a digital code, or other signal, as its output. In that case, the output signal may be processed by the vehicle's computer, and the temperature data fed to the vehicle's data bus. The thermal control of the swirl-vane orientation apparatus may then include a data-bus reader, and a transducer for converting the temperature data into mechanical movement.

[0088] The coolant temperature sensor can be indirect. The sensor might measure engine-oil temperature directly, for example. In fact, measuring the oil temperature can sometimes lead to greater efficiencies; studies have indicated that controlling the oil temperature can give even greater improvements in efficiency than controlling the cooling-coolant temperature - insofar as the two effects can be separated. It should be understood that a sensor that is so placed as to measure directly the engine-oil temperature, is still, for the purposes of the invention, a sensor for measuring the temperature of the engine coolant. Similarly, if the temperature sensor were to be so placed as to measure directly the temperature of the metal of the engine block, that would still, for the purposes of the invention, be a sensor for measuring the temperature of the engine coolant.

[0089] Alternatively, the designer can arrange that the flowrate that is thermally controlled can be the flowrate of the oil, rather than (or as well as) the flowrate of the coolant. In this context, it should be understood that the expression coolant includes the engine oil, in the case where the oil is being circulated (i.e pumped) around the engine, and where, during operation of the engine, substantial heat transfer takes place between the engine

components and the oil.

[0090] One of the advantageous aspects of the swirl-vane technology is the improved resistance to cavitation in the pump impeller. Cavitation arises when the pressure of the fluid actually in contact with the impeller blades is below the vapour pressure at a given temperature, whereby a cavity of vapour is formed, contiguous to the impeller blades. Cavitation not only spoils the efficiency of the pump, but can lead to vibration, erosion, and other pump problems.

[0091] Cavitation in the blades of a pump, if it occurs, can cause a significant drop-off in the volumetric flowrate of the liquid passing through the pump. In an automotive cooling system, pushing back the onset of cavitation can be very important.

[0092] Electrically-driven pumps go well with electronic data processing. The combination makes it simple to optimise the output of the pump (for maximum flowrate, or maximum efficiency, etc, as conditions may require), over the speed range of the engine, and over the temperature and other operating ranges of the engine. As discussed, although the speed of an electric motor may be controlled electrically (at least just from the standpoint of accuracy of control), and thus it is easy to tailor the pump output to system requirements, still, controlling pump output by controlling the orientation of the swirl-vanes in accordance with temperature may be a better compromise between cost and performance. By being able to tailor both the swirl-vanes orientation, and the pump speed, in accordance with temperature (and other parameters), engine coolant temperatures can be kept very close to optimal under almost all conditions.

[0093] But even when the coolant pump is mechanically driven, from the engine, as described in relation to the illustrated embodiments, deriving temperature sensor data electronically, from the data bus, can give a quicker response than using a mechanical thermostat unit.

[0094] When the temperature sensor data is in the form of an electronic signal on the data bus, the designer may arrange for the swirl-vanes to be orientated by means of a computer-controlled stepper-motor, or servo, which again is in keeping with the trend towards greater electronic control.

[0095] When the temperature information is in the form of an electronic signal on the data bus, the designer is able to also arrange to coordinate the radiator cooling fan motor with the

pump speed, in order to realise better overall efficiencies in the coolant system. The designer's overall aim is (usually) to maintain optimal engine temperature, while expending a minimum amount of energy to run the coolant system.

[0096] Thus, when the degree of swirl-vane biassing is controlled by the temperature of the coolant, as engine-monitoring becomes more sophisticated, so it becomes more possible for the volumetric flowrate produced by the coolant pump to be truly optimised to the thermal conditions. The desired effect is that engine temperature can be controlled within tighter limits, and that as little energy as possible be drawn by the pump.

[0097] When the temperature sensor signals are electronic, generally there is no mechanical connection between the structure of the temperature sensor and the structure that moves the vanes. Rather, the signal controls a servo, and it is the servo that provides the mechanical drive to re-orientate the swirl-vanes.

[0098] When the coolant pump is driven by an electric motor, it can be beneficial for the designer to specify that the motor run at constant speed. However, constant speed is not essential. There is a trend, in electric motors, to commutate electric motors electronically, not mechanically. The motor speed will be on the data bus, whereby it becomes a relatively simply matter to relate motor speed to coolant temperature, as well as to relate swirl-vane orientation to coolant temperature.

[099] In the traditional simple type of automotive thermostat, a desideratum has been that the thermostat should remain closed up to a temperature close to 195 deg-F, but beyond that the thermostat should go fully open. In practice, opening does not take place suddenly, once the set temperature is reached; rather, a conventional simple thermostat might be set to start to open at a temperature of e.g 180 deg-F, and opening is not complete until about 200 deg-F.

[0100] Fig 12 is a graph showing the characteristic of the thermostat 235, which is of the type known as a double-break thermostat. Here, the y-axis represents the extension of the stem of the thermostat bulb unit for the different temperatures as plotted on the x-axis. The stem starts to move at about 210 deg-F, and moves then at quite a high rate, whereby the stem has extended 0.14 inches at 220 deg-F. After that, the stem moves at the very slow rate of about 0.01 inches per ten degrees rise, whereby for the next 26 degrees, i.e up to 235 deg-F, the stem moves only a further 0.05 inches. Beyond 235 deg-F, the stem moves at the rather greater rate of 0.1 inches per ten degrees rise.

[0101] It is recognised that the double-break thermostat bulb unit is very well suited especially to the embodiment described herein where the function conventionally performed by the engine radiator thermostat valve is performed by the swirl-vanes. The initial movement of the stem takes place relatively suddenly, and the movement of the stem is of sufficient magnitude as can easily be harnessed to move the swirl-vanes from the closed position to the position of minimum flow-boost. After that, the change in the orientation of the swirl-vanes per degree of coolant temperature is very small, whereby the swirl-vanes remain more or less stationary in the minimum flow-boost orientation until the temperature reaches about 235 deg-F. Beyond that, the swirl-vanes start to change orientation at a more rapid rate, up to the maximum flow-boost position, which occurs at about 245 deg-F.

[0102] With a double-break thermostat, the designer can specify the change points to be at particular temperatures, as required, to suit the characteristics of particular engines. The double-break thermostat not only provides different rates of movement of the stem (i.e rate, as measured in millimetres-per-degree) over different temperature ranges, but also provides the designer with the flexibility to specify the temperatures at which the rate changes, to suit the particular case. Initially, in Fig 12, the swirl-vanes move from closed to partway-open rapidly, just as the coolant reaches warmed-up temperature. It is a simple matter to tailor the rate of stem movement, i.e the millimetres of movement per C-degree, to be more or less exactly what is required, i.e: a small opening at a rapid rate at first (to open the radiator port just as the coolant goes from cold to warmed-up); then slow rate (to leave the swirl-vanes more or less unchanged as the coolant goes from just-warmed-up to hot); then a rapid rate again, though not so rapid as initially, to effect movement of the swirl-vanes, to give a large flow boost, as the coolant goes from hot to very hot.

[0103] Thus, the double-break mechanical thermostat (known per se) is of considerable benefit when used in the kind of coolant pump as described, where the stem movement of just one thermostat is used both to effect the close/open movement of the radiator-port-closer, and to effect the progressive flow-control and flow-boost movement of the variable swirl-vanes.

[0104] Similarly, in installations where the temperature sensing is done with electronic sensors, and the movement of the radiator-port-closer and of the swirl-vanes is done by e.g computer-controlled stepper motors, it is a simple matter in that case, too, for the designer to ensure that the movements of the components are co-ordinated in the most efficacious manner.

[0105] In the case of a particular vehicle, fully loaded, travelling uphill on a hot day, the coolant flowrate might need to be, for example, 100 litres per minute. On the other hand, the same vehicle, cold day, downhill, might need less than a tenth of that flowrate. It may be expected that the thermally-actuated swirl-vanes, as described herein, when properly designed, can enable at least most of that difference to be achieved. However, when the swirl-vanes are compromised by combining the function also of opening/closing the radiator port, it may be expected that, while such very large differences in flowrate cannot be achieved, still the cost savings arising from the fewer components make the combined-action swirl-vanes worthwhile.

[0106] Ideally, the thermal-actuation of the swirl-vanes, as described herein can, at least notionally, provide a coolant flowrate that, under all operating conditions, is effective to keep engine temperature optimum, and to be so by providing just the flowrate required, without compromising or wasting excessive flowrates and pressures. Combining the thermally-actuated radiator-port-closer with the thermally-actuated swirl-vanes is a compromise, which might sometimes make the ideal rather less obtainable than when the two thermal actuators are separate and independent; but on the other hand, using a common thermal actuator for both tasks gives a considerable cost saving, compared with using two independent thermal actuators.

[0107] In other words, providing thermally-actuated swirl-vanes to adjust coolant flow enables large economies to be made overall to a vehicle's cooling system. This is true, especially, when compared with a system in which the radiator port is opened/closed by means of its own independent thermostat. But combining the thermal-actuators is a direct cost saving, which at the same time enables at least a portion of those overall economies to be made.

[0108] In this specification, it has been stressed that the swirl-vanes and the radiator-port-closer be located inside the pumping-chamber. The expression "located inside the pumping-chamber" will now be considered. The pumping-chamber is the structure that houses the impeller, and which constrains flow through the impeller, and is the structure that extends upstream (and downstream) from the impeller for sufficient distance that the flow within the pumping-chamber has a rotational component of velocity induced by the impeller. That is to say: flow outside or beyond the pumping-chamber has no, or no substantial, rotational component of velocity induced by the impeller.

[0109] (The said rotational component of velocity is, as mentioned, induced by the rotary motion of the impeller itself, and should be distinguished from the swirl motion as described in this specification. The swirl motion may be imparted to the flow either in the same rotational sense as, or in the opposite rotational sense to, the rotation of the impeller; the rotational component of velocity induced by the rotation of the impeller is always, of course, in the same sense as the rotation of the impeller. Given that the swirl-vanes are located inside the pumping-chamber, it should be understood that the impeller-induced rotational component of velocity is the component that would be present if the swirl-vanes were not present.)

[0110] The coolant in an automotive engine passes through many passageways, vaults, chambers, hoses, pipes, etc, as it flows around the cooling system. The whole flow divides and recombines many times. Generally, the smallest cross-sectional area through which the whole flow passes (and where its velocity is greatest) is the cross-sectional flow area of the impeller itself, i.e the minimum cross-sectional flow area through the impeller blades, being min-A sq.mm. The squareroot of min-A sq.mm is min-D mm.

[0111] Ideally, the pumping-chamber should be designed to constrain the liquid to pass through a cross-sectional area that decreases gradually and progressively as the liquid approaches the impeller, and increases gradually and progressively as the liquid leaves the impeller. Of course, usually it is not possible to design a pumping-chamber solely for maximum flow efficiency. In some cases, design constraints might mean the pumping-chamber only extends e.g ½ x min-D mm upstream /downstream of the impeller. (Again, the pumping-chamber is the portion of the flow-constraining walls that conducts flow through the impeller, in which the flow has a substantial rotational component of velocity.) Even when the pumping-chamber is designed with flow efficiency as the main criterion, the rotational component of the velocity of the flowing liquid only extends only a few tens of millimetres upstream /downstream from the impeller. For the purposes of the invention, i.e the preferred form of the invention, it can be regarded that no portion of the flow of liquid that lies more than about 1½ x min-D or 2 x min-D millimetres from the impeller, as measured along the direction of flow, can be within the pumping-chamber. That is to say: in the preferred invention, any rotational component of velocity present in portions of the flow lying more than about 1½ x min-D or 2 x min-D millimetres from the impeller would not be substantial. In the invention, preferably the radiator-port-closer lies within the pumping-chamber, or at least a substantial portion of the structure of the radiator-port-closer should lie within the pumping-chamber. If there are e.g bends in the housing walls, or other discontinuities, that prevent

impeller-induced rotation from being transmitted therebeyond, portions of the walls lying beyond the bends or discontinuities would not be portions of the pumping-chamber.

[0112] Cooling systems differ in different engine designs, especially as to the manner in which the coolant circulates around the engine when the radiator port is closed, and as to how the heater is brought into the circuit. The layout invariably includes by-pass conduits, through which a small flowrate of coolant can circulate through the engine when the coolant is cold and the main flow through the radiator is blocked. These by-pass conduits may include e.g pressure-sensitive check-valves. The arrangement of the by-pass conduits may be such that the cold by-pass flow through the engine simply passes also through the heater. Or it may be such that the flow, when the coolant is cold, passes only through the engine and not through the heater, and the flow through the heater only commences at a temperature such that the coolant has warmed up a little (but not enough to open the radiator port). Or it may be arranged that, when the coolant is very cold, there is no flow even through the engine, until the coolant has warmed up a little. In that case, coolant starts to circulate through the engine at a first temperature, then through the heater at a higher temperature, then through the radiator at a higher temperature still; this system of course requires more sophisticated temperature sensing (and more sophisticated transducing into mechanical movement) than can be supplied by a single conventional thermostat.

[0113] The present invention is generally applicable, whatever particular arrangement is provided for the cold by-pass circulation. The invention aims to provide a cost-effective manner of combining thermal modulation of the hot main circulation using the swirl-vanes, with a manner of routing the cold by-pass flow through the engine and pump while flow through the radiator is blocked. The designer will naturally adapt the particular layout of passageways and conduits to suit the particular design of cold by-pass circulation. It may be arranged that cold flow through the heater is circulated therethrough by means of a separate pump - i.e separate from the main coolant circulation pump; in that case, provided the cold by-pass flow through the engine still passes through the main coolant pump, the invention may be applied. Indeed, the invention may be applied to engine coolant circulation systems that do not have a heater at all. In the case of an engine that uses two separate coolant circulation pumps, one to handle the cold by-pass circulation and the other to handle the hot main circulation, the present invention would not apply.